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LNG Propane Compressor Performance Prediction and Large scale Test Validation

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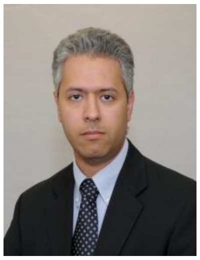

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ABSTRACT

This paper presents results of the large scale testing of the first two stages, including incoming sidestream flow of a LNG propane compressor. The large scale test was performed at 90% scale of the production propane compressor. The four section propane compressor contains five stages and three incoming sidestreams. The last four stages are of the same impeller family. Therefore, only the first two stages were tested. This paper will describe the complete process from initial selection and design to the scaled testing. Performance requirements for this compressor were tighter than standard API-617 guarantees. The limitations required for the test were +2% for power, -1% for speed and +3% for head at design flow, ±2% for sidestream pressures and -2% for surge/stall and overload/choke margins. The design process of the stationaries (IGV, diffuser and return channel vanes) and the sidestream plenum and mixing section will be described. Computational Fluid Dynamic (CFD) analyses were used to size the plenum and mixing section in order to reduce losses and ensure a uniform flow distribution for the subsequent stage. Internal stage performance (IGV, impeller, diffuser, and return channel) predictions were also generated using CFD analysis tools. A newly developed tool (presented at 2012 Texas A&M Turbomachinery Symposium) was used to predict the sectional performance and flange pressures. Due to the tighter performance limitations, the impellers were single piece fabricated and an advanced laser 3D scanning method was used to ensure components met specified manufacturing tolerances. This paper will describe the test rig, instrumentation and test procedures employed. The test was completed for two full (overload to surge) speed lines at the design flow ratio. Test results will be compared against the CFD predictions for stage and sectional performance. Conclusions and recommendations will be presented, including ASME PTC10 guidelines suggested changes.

INTRODUCTION

There are many different process cycles that can be selected to convert natural gas to liquid form. These include Air Products AP-M™ (single mixed refrigerant), AP-C3MR™ and AP-X™, ConocoPhillips Optimized Cascade® process, Shell's Double Mixed Refrigerant (DMR) and Parallel Mixed Refrigerant (PMR), and the Black and Veatch PRICO® process. All refrigeration cycles require compression equipment but the compression requirements for each process vary, because each cycle requires different refrigerant gases (propane, ethane, butane, ethylene, methane, and nitrogen).

The resulting variation in gas properties, ambient conditions and feedstock composition leads to different compressor requirements. The common requirements for all process cycles are high compressor efficiency and accurate prediction of the compressor performance. Compressor efficiency has a direct impact on LNG production for a fixed amount of power, while accurate performance prediction minimizes changes after compressor testing and reduces design margins for plant equipment dependent on the compressor.

BACKGROUND

The Original Equipment Manufacturer (OEM) initiated a multi-year effort to accurately predict compressor performance while simultaneously extending the operating range and efficiency of main refrigeration compressors. This effort required improvements to the existing design process, including improved correlation of three dimensional (3D) or CFD tools used in the design process (Kowalski et al [1]). Additionally, improvements were made to the prediction of the flange pressure for compressors with incoming sidestreams, which are typically used for propane refrigeration compressors in the C3MR process (Koch et al [2], Fakhri et al [3]).

This paper will describe an effort undertaken with client cooperation to validate the entire design and prediction system, including impeller development, stage performance prediction, and sidestream prediction. For this validation effort a propane compressor with sidestreams (typical of a C3MR process) was selected using conditions from an LNG plant design study being supported by the OEM. The intent of this study was to validate that the OEM tools could design and accurately predict the performance within the stringent tolerances requested by the client.

The standard power guarantee for API-617 [4] is +4% with the stability being within 10%. The accuracy requirements requested by the client were of +2/-0% for power, -1% for speed and +3/-0% for head at design flow, ±2% for sidestream pressures and -2% for surge/stall and overload/choke margins.

The LNG refrigeration propane compressor that was analyzed through this test validation was comprised of four sections, five stages and three incoming sidestreams (see layout of the unit in Figure 1). Section one includes a medium head, high flow coefficient and high Mach impeller in stage one. Sections two through four include low head, high flow coefficient and high Mach impellers in stages two through five. Since the last four stages are of the same impeller family, low head, high flow coefficient and high Mach, the validation test was limited to stage 2. Therefore, the validation test would represent stages 1 and 2 of the production unit.

In order to meet the market requirement for better performance and operating range for high Mach applications, the OEM embarked several years ago on the development of various high flow coefficient stages operating at different head levels. The stages were then validated in the OEM subscale test rig and the excellent agreement between test and CFD prediction was presented in a previous paper (Kowalski, et. al [1]).

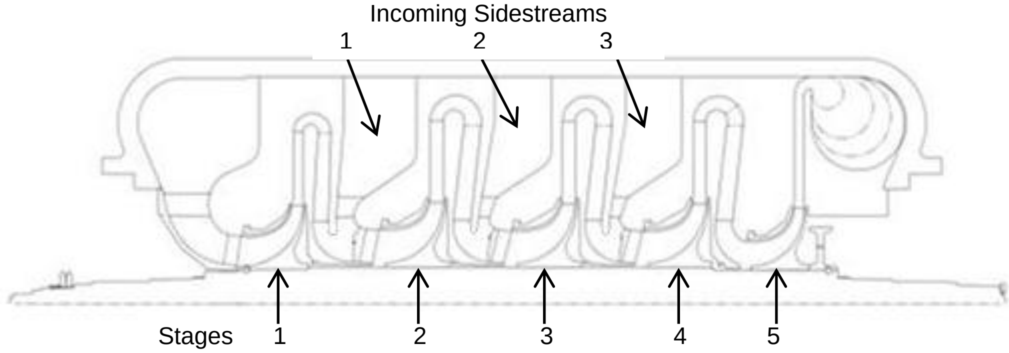

Figure1: Production compressor layout.

The validation test for this propane refrigeration unit was completed at the OEM's LNG full scale test rig. The large scale test was performed at 90% scale of the production propane compressor. This eliminates the impact of frictional scaling effects and ensures a valid performance prediction. The validation testing consists of the first two stages of the production unit, including the first incoming sidestream flow (see layout of the test unit in Figure 2). A third stage impeller (booster stage) was added to augment the pressure rise. This is needed to generate a complete operating map for stages 1 and 2 (overload to surge/stall).

Performance prediction for multi-stage centrifugal compressors with multiple sidestreams is quite challenging. To predict accurate sectional or flange-to-flange performance, pressure, temperature and flow conditions at each of these incoming flow streams must be met within stringent pressure and power tolerances. For each section, the mixing of the mainstream flow and the sidestream flow must be fully understood since this affects sidestream flange pressure and to also ensure that the following stage is correctly sized and produces the necessary performance. The work of Sorokes et al. [6] has led to a greater comprehension of the complex flows at the sidestream mixing location. However, this work did not address the flow physics that determine the static pressure in the mixing section, which, in turn, sets the flange pressure level. The study by Hardin [7] describes a one-dimensional method that determines how the flow at the mixing location and, therefore, the local static pressure is impacted by the local flow curvature. A study by Koch et al. [2] was conducted to predict the downstream total pressure and compare the analytical results to measured test data on two different geometries. This paper recommended a revised methodology to accurately predict mixing zone conditions by taking the local curvatures into account. It was validated using test results and was found to be within ± 1.5% error for the total pressure at the flange. The sidestream modeling used in this paper was developed by Fakhri et al [3]. This methodology has been validated via comparison with subscale rig test data. The predictions demonstrated excellent agreement with the experimental results.

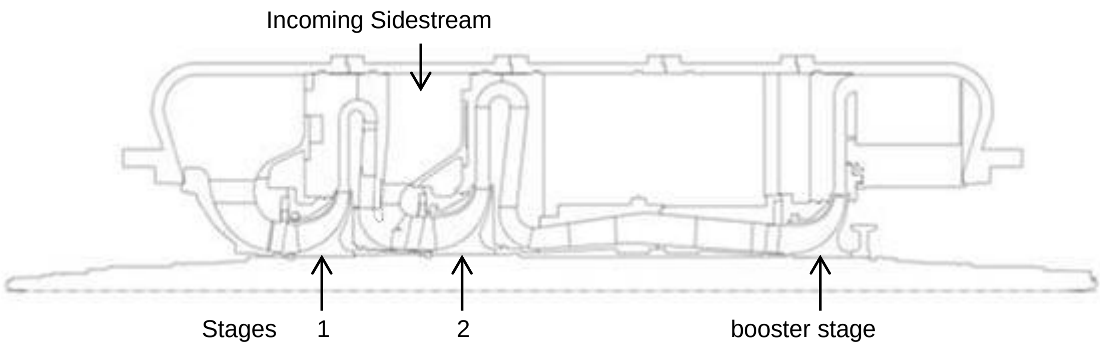

Figure2: Large scale test compressor layout.

Acceptance testing of industrial centrifugal compressors is governed by specification API-617 [4] and ASME Performance Test Code (PTC-10 1997) [5]. The ASME code provides stringent guidelines that must be met to achieve test similitude. The primary parameters for similitude are impeller tip Mach number and the ratio of volume flow entering and discharging the compressor (volume ratio). For applications where sidestream flows are present, the code stipulates limits on the ratio of flow entering the sidestream versus the mainstream flow exiting the upstream section. The acceptable variation in volume flow ratio for intermediate sections of a compressor with multiple sidestreams is ±10% (±5% for first section). The effect of varying flow ratio in a sidestream has a potentially significant impact on the flange total pressure and hence the sectional performance. Many different parameters affect sidestream sectional performance and of particular importance are: the frictional losses in the sidestream geometry (flange, plenum and mixing section geometries); the static pressure changes due to local curvature and mean velocity; and the flow ratio between the sidestream and mainstream flows. The flow ratio is determined during the test. Flow ratio, per the ASME code (PTC-10 1997), is defined as the ratio of the volume flow rate at the sidestream flange to the volume flow rate at the exit of the preceding stage. The effect of varying flow ratio within the specified ASME tolerances could result in head for the section being off by as much as 5 to 9 percent, while the pressure at the flange will be off by the same amount as the flow ratio is allowed to change. Other than affecting the sectional performances, it is also important to note that variance in flow ratio impacts the velocity levels where the two streams merge. Significant variation in the velocity profile of flow entering a downstream impeller changes the incidence on the blade leading edge of that impeller. This leads to a change in the stage (internal) performance.

DESCRIPTION OF THE COMPRESSOR DESIGN

Stage design

The production propane compressor uses newly designed high flow ($\varphi > 0.1000$), high Mach number impellers (machine Mach > 1 and shroud relative inlet Mach number > 0.9). The primary intent of these new designs was to increase the stage efficiency and overall range. The new stages were developed using a typical design methodology; i.e., a blend of 1-D, 2-D and 3-D analytical methods.

The full-inducers, arbitrary-bladed impellers were developed for use in the first, intermediate or last stage of a multi-stage, beam-style centrifugal compressor. The new stages were required to operate over a machine Mach number range of 0.85 to 1.20. The stage also included an upstream vaned inlet guide and a vaneless diffuser, return bend and return channel (or de-swirl cascade). The parallel wall diffuser width was set such that the flow angle did not violate any of the widely accepted stall avoidance criteria (i.e., Senoo [8, 9], Kobayashi-Nishida [10]). The return channel system was also developed using well-established design guidelines, ensuring proper leading edge incidence, passage area distribution, vane shape, etc. to avoid any untoward flow anomalies.

Sidestream design

The three sidestreams for the production propane compressor were designed following a detailed design approach:

1. For clarification on the terminology employed, the reader can refer to figure 3(a). The meridional contour of the "mainstream side" and "sidestream side" of the tail was sized based on the amount of flow coming from the upstream stage and sidestream respectively, and to provide a uniform flow velocity profile at the impeller eye of the downstream stage. Passage area distributions through the "mainstream side" and "sidestream side" were adjusted to obtain adequate flow acceleration. CFD analysis on the mixing section was run (using the methodology described in the section *Analytical set-up*) and compared to the equivalent mixing section without tail (i.e., Figure 3(b)). The design of the sidestream mixing section was considered successful when the variation in velocity profile at the impeller eye, meridional static pressure contour and other parameters were negligible compared to the mixing section without tail.

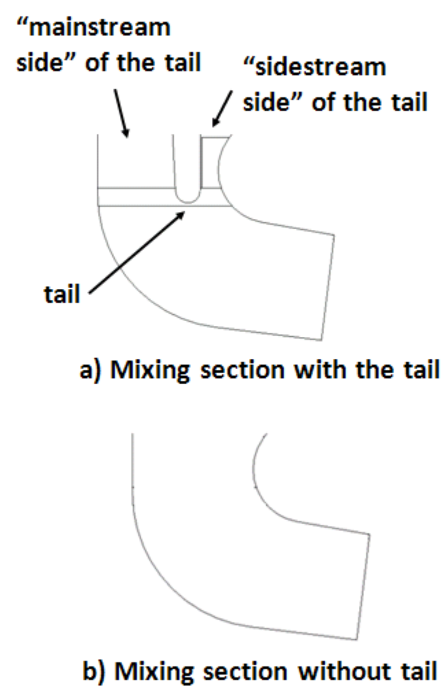

Figure 3: Mixing section geometries: (a) with the tail, (b) without the tail.

2. The sidestream flange and plenum were designed following OEM guidelines. The flange diameter was selected such that the Mach number at the inlet flange was within acceptable levels to minimize frictional losses. The plenum area was sized to allow for adequate collection of flow. The meridional contour was shaped to minimize

recirculation zones from developing in the plenum, to minimize frictional losses, and to smoothly guide the flow into the collector vanes. A set of vanes was inserted into the plenum for mechanical support and to uniformly distribute the flow. Opposite to the sidestream flange, a seagull was added to guide the flow towards the scoop vanes. Figure 5(b) illustrates the plenum design.

3. A set of vanes was included into the plenum. The vanes were composed of a curved portion morphing into a radial portion to remove swirl. The vane length and number was selected to satisfy OEM solidity criteria. The radius of curvature of the collector vanes was adjusted such that the vanes aligned with the incoming flow. The collector vanes were symmetrical relative to the vertical axis.

CFD analysis (using the methodology described in the section *Analytical set-up*) was run on the full 360 degree plenum model. The domain included a sector model of the return bend and return channel of the upstream stage (i.e., figure 5) such that the effects of mixing the core flow and sidestream flow were assessed. Several factors were monitored to verify that the sidestream design was adapted to the operating conditions: uniform velocity profile radially and circumferentially at the impeller eye, swirl angle at the impeller eye, pressure, Mach number, temperature contours at the mixing section, sidestream loss coefficient and flow incidence at the scoop vane leading edge. Figure 4 displays the streamlines colored by Mach number across the sidestream and upstream stage.

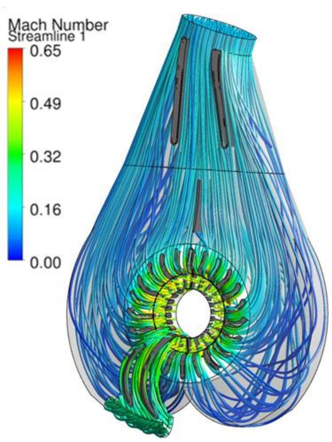

Figure 4: Sidestream analysis - streamlines colored by Mach number.

ANALYTICAL SET-UP

All analyses described in this paper were conducted using the commercially available solver, ANSYS/CFX-12.1 and 13.1. CFD analysis was performed on the main inlet, the sector models of each stage, sidestreams, and the last stage with the discharge volute for both the propane production compressor and the large scale validation compressor. Further, all analyses were done at their respective scales.

Computational Domain

Figures 5, 6 and 7 display the computational domains used to perform the CFD analysis on the main inlet, sidestream, stage analysis and the last stage with the discharge volute. The computational domain for the main inlet was composed of the plenum and the vanes. The geometry for the sidestream analysis included the sidestream plenum and vanes, and a sector model of the return bend and return channel of the upstream stage to assess the impact of mixing the mainstream with the sidestream flow. The computational domain for the stage analysis was a sector or “pie-slice” model that included the upstream inlet guide (i.e., IGV), impeller, diffuser, return bend, return channel, and exit section if there was no sidestream upstream the stage. The upstream inlet guide was replaced by a sector model of the mixing section if a sidestream was located upstream of the stage. The computational domain for the last stage was composed of IGV, impeller, diffuser, and the discharge volute. For the large scale validation test compressor, CFD analysis from the bridge over inlet to the last stage diffuser exit was run as well.

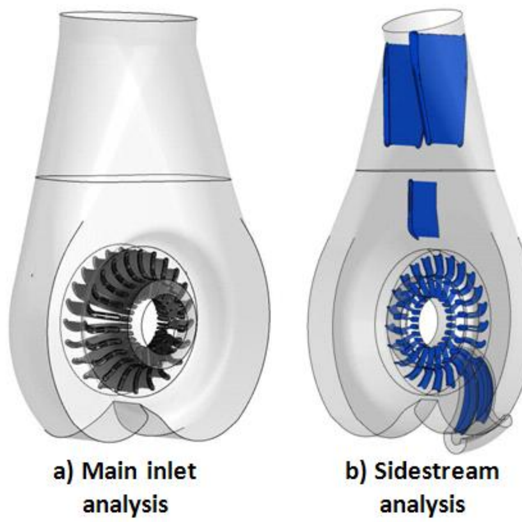

Figure 5: CFD domain of (a) the main inlet, (b) sidestream.

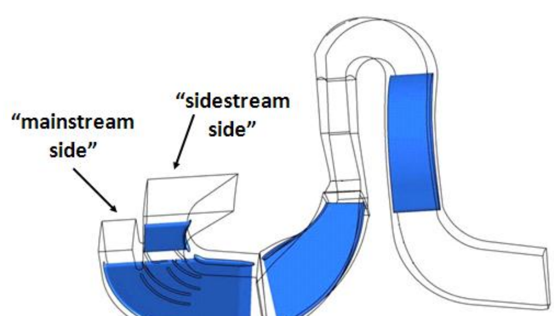

Figure 6: CFD domain of the stage analysis with the mixing section upstream.

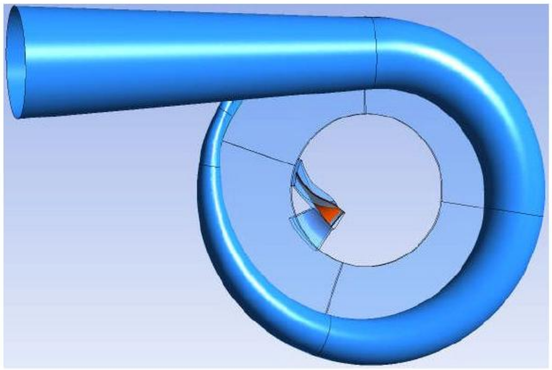

Figure 7: CFD domain of the last stage including the discharge volute.

- *Computational Mesh*

All components were meshed using an unstructured grid. The meshes contained sufficient elements to ensure mesh independent results. Inflation layers along the walls and the vanes/blades of each component were added to capture any flow instabilities that developed in the boundary layers. Great care was taken in the sizing of the inflation layers and the y+ to adequately envelop the boundary layer.

- *Boundary conditions*

All analysis was run assuming steady state conditions. The interfaces between the components were modeled using (1) a "stage" or circumferentially averaged interface if one of the components was a sector model or (2) a "general connection" interface if both components were full 360. The k-ε turbulence model and a high-resolution discretization scheme were applied as the OEM has had good success using these approaches where good correlations between the test results and the CFD analysis have been obtained. The convergence criteria for the steady state CFD analysis were a maximum residual below 1E-03 and convergence in critical parameters, such as total pressure, Mach number and total temperature. These guidelines ensure consistency between the various results and adhere to recommended practice.

The boundary conditions imposed for the CFD analysis were total pressure and total temperature at the inlet and mass flow at the outlet of the domain for analysis where the computational domain contained only one inlet (i.e., main inlet, stage analysis without sidestream upstream or last stage analysis). For the sidestream analysis, the boundary conditions were: total pressure, total temperature and flow direction (data extracted from the stage analysis) at the return bend inlet of the upstream stage; total temperature and mass flow at the sidestream flange inlet; and mass flow at the outlet of the domain. Note the mass flow at the sidestream flange inlet was adjusted to satisfy the sidestream flow function ratio (i.e., Equation 1). A similar set-up for the boundary conditions was applied to the stage analysis, including the mixing section, with the exception of the flow direction considered normal to the boundary and the mass flow at the sidestream inlet, being independent of the sidestream flow function. To improve the convergence of the simulations near the overload condition, static pressure was used as boundary conditions for the stage analysis. The values of inlet total pressure, total temperature, flow function ratio, outlet mass flow, and speed were identical to the operating conditions on the field for the production propane compressor or tested for the large scale validation compressor. The CFD analysis was run using the same gas as during the test; i.e., propane for the propane production compressor and R134A for the large scale validation compressor. The Redlich-Kwong equation of state was used to estimate the gas characteristics in the analyses.

The overall flow field in the main inlet, stage and sidestream were assessed from overload to surge. For the CFD analysis, the surge point was defined at the flow rate where the stage head coefficient had peaked or until the critical convergence parameters fluctuated beyond acceptable limits. Based on the OEM's experience, the above criteria is indicative of the flow rate at or near which the stage will stall on test.

OVERALL TEST VEHICLE DESCRIPTION

The tests were conducted in the OEM's full scale test rig, which was built to validate the aerodynamics and mechanical performance of large scale compressors. To accommodate various compressor arrangements (i.e., multi-stage, back-to-back compressor and/or intermediate sidestreams), the compressor casing is composed of a series of rings as seen in Figure 8. The full scale rig can accommodate impellers up to 60 in (1525 mm) in diameter.

Figure 8: Segmented Casing on test stand.

The tested large scale validation compressor was a three-stage machine with an incoming sidestream between the first and second stage (as shown in figure 2). Of interest for this project was to assess the performance of the first two sections of the production machine. The third stage was added to the compressor to overcome the test loop system resistance. A bridge over connected the second to the third stage.

The rotordynamics of the large scale test vehicle were validated in a previous client test, which included two sidestreams and four stages. The shaft and bearings were re-used for this test. As provision for potential flow and/or performance adjustments, moveable inlet guide vanes (IGV) were installed in all stages. The vane setting angle was controlled by actuators that were mounted on the outside compressor casing. The system actuator vane setting angle was calibrated such that the angular change in the vane setting angle matches to a linear motion of the actuator.

The compressor was driven through a gearbox by a 22.37 MW (30,000 HP) steam turbine. The compressor is installed in a closed-loop system similar to the schematic shown in Figure 9. Available test gases include nitrogen, carbon dioxide, R-134A refrigerant, and, if necessary, helium-nitrogen mixtures.

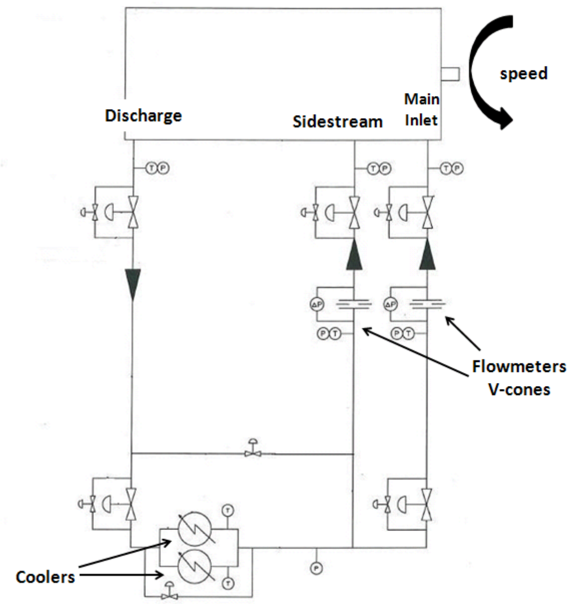

Figure 9: Schematic of test loop with main components.

The test was run with R-134A as the gas medium and was conducted on the first and second sections of the compressor. First stage performance was assessed from flange-to-flange, while the second stage performance was evaluated from flange-to-return channel exit. Each stage was tested at three operating conditions: (1) design machine Mach number and radial IGV; (2) 95% design machine Mach number and radial IGV; and (3) design machine Mach number and against IGV (i.e., against IGV refers to the moveable IGV being rotated in the direction opposite the compressor rotation). The machine Mach numbers for the first and second stages are 1.13 and 1.14, respectively. The test was conducted in accordance with ASME PTC-10 [5] guidelines for similitude between the production propane compressor and the large scale validation compressor. ASME PTC-10 defines the acceptable level of deviation regarding the volume reduction, inlet volume flow, machine Mach number, and Reynolds number between the field and test operating conditions. Note that for sidestream compressors, ASME PTC-10 requires sectional performance to be evaluated from flange-to-flange. The effect of the second sidestream on the flange-to-flange performance of the second section will be predicted using an in-house tool [3]. The code defines the flow proportion between the mainstream flow and the sidestream flow as a ratio of volume. To account for the difference in densities [11] between the test gas and the specified gas, the volume term was replaced by: $M * \frac{\sqrt{R.T.Z}}{P}$. The flow function ratio was defined according to Equation 1; as the ratio of the side stream flow divided by the flow from the return channel exit of the previous stage. Fakhri et al. [3] demonstrated the sectional performance was highly sensitive to the flow function ratio. For this test, the deviation in flow function ratio was maintained within +/-3%, with the possible exception of flows near choke. ASME PTC-10 code recommends the tolerance in flow function ratio to be within +/-10%.

$$FF_{ratio} = \frac{\left(\frac{M* \sqrt{R.T.Z}}{144*P}\right)_{SS}}{\left(\frac{M* \sqrt{R.T.Z}}{144*P}\right)_{RC\ exit}}$$

Equation 1 : Definition of the flow function ratio

M = mass flow ($lbm.min^{-1}$)
R = specific gas constant ($ft.lbf.lbm^{-1}.^{o}R^{-1}$)
T = total temperature (^{o}R)
Z = compressibility factor
P = total pressure (psia)
RC exit = return channel exit of stage 1
SS = sidestream flange inlet

For each operating condition, a full-speed line of data was taken on both stages 1 and 2; typically eight to twelve thermodynamically settled flow points from overload to stall/surge. Further, the data acquisition system captured performance data every four seconds, accumulating so-called “transition data” that provide additional insight into the compressor operation between settled points.

The compressor was extensively instrumented internally, as shown in figure 10, with half-shielded thermocouples, Kiel head pressure probes, 5-hole probes, static pressure taps, five elements pressure and temperature rakes, and dynamic pressure transducers. In addition, proximity probes were used to monitor rotor vibrations and validate the mechanical behavior of the unit. The dynamic pressure transducers (which detect pressure fluctuations) at the diffuser inlet and exit have proven to be critical in the detection of surge and stall phenomena. To acquire data at each flange, eight thermocouples and Kiel probes were inserted at the main inlet, sidestream and discharge flanges.

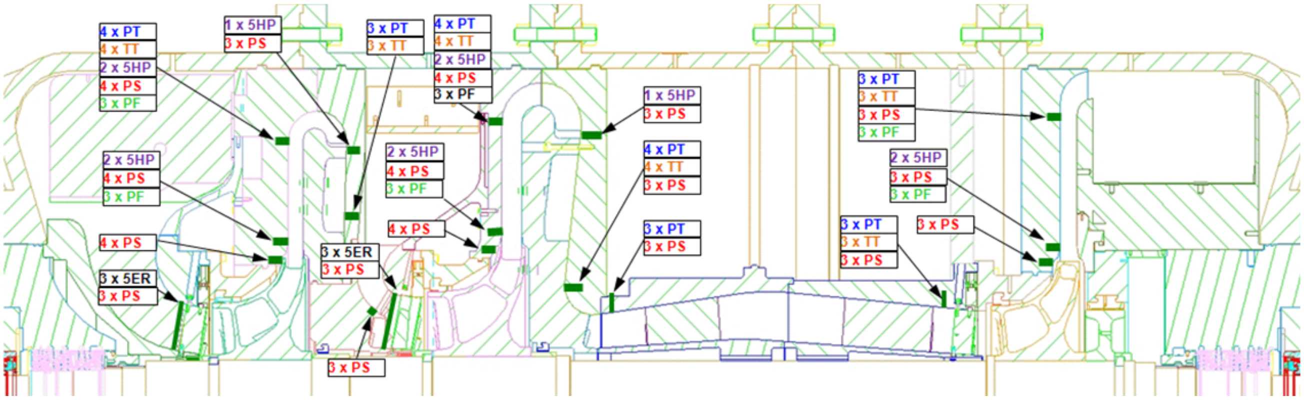

Figure 10: Internal instrumentation for the large scale validation compressor

For this test program, there were three dynamic pressure transducers mounted both at the diffuser inlet and at the diffuser exit of each stage. The dynamic probes were distributed circumferentially to detect stall cells as well as the number of cells and their direction of rotation. The probes were selected to be more sensitive to lower frequencies; i.e., those typical of diffuser and/or impeller rotating stall.

There were two 5-hole probes at the diffuser inlet, two at the diffuser exit and one at the entrance of the return channel for the first two stages. The 5-hole probes measure static pressure, total pressure and flow angle at the probe location.

Several observers monitored the vibration and dynamic pressure transducer signals as well as the real-time display of the flow angles (and other aerodynamic parameters) to take note of any interesting phenomena that occurred during the testing. In short, every effort was made to ensure that all critical data was captured during the run.

Experimentally, surge was defined as the flow rate at which the head coefficient decreased or sub-synchronous frequencies consistent with rotating stall appeared.

TEST RESULTS

Extensive instrumentation placed at critical locations within the test rig allowed comprehensive data to be collected to determine internal (stage) and external (sectional) performance. To optimize the performance of the propane production machine, the large scale validation compressor was run at off-design conditions (i.e., 95% design Mach number and against IGV) in addition to the design conditions.

For accurate sectional performance predictions of compressors with multiple incoming sidestreams, it is critical that the following criteria be predicted accurately:

1. Internal (stage) performance.
2. Sidestream (mixing section, plenum and flange) performance.

Sidestream sectional performance (flange-to-flange polytropic efficiency and head coefficient) are generally very sensitive to the flow ratio between the sidestream (flange) and mainstream (internal) flows. Errors are further compounded by inaccurate stage performance predictions, as this leads to incorrect stage sizing/selection. Since the discharge pressure of the stage preceding the sidestream is instrumental in setting the sidestream flange pressure, it is essential that the stage discharge pressure be predicted accurately. For this reason, CFD analyses were performed on each stage prior to testing the compressor.

Test Results vs CFD prediction at Design Speed

In the past, the OEM has obtained good agreement between test measurements and CFD analyses, especially for high Mach application stages [1]. The test results, at design speed, of the large scale validation compressor were compared with the corresponding CFD prediction for the two stages.

Figures 11 and 12 present the normalized overlays of the test performance with the predicted performance at the design U_2/A_0 speed (1.13 and 1.14 for stages 1 and 2 respectively).

For both stages, the polytropic efficiency and head coefficient from the test agreed well with the predicted CFD performance. The test results showed a better surge margin for both stages relative to the predictions. The surge margin as predicted for stage 1 (Figure 11), came out to be close to the surge margin from the test. For stage 2 (Figure 12), the CFD under predicted the surge margin. It is important to note that surge and stall, being non-steady phenomena, are difficult to predict using steady-state CFD. Transient simulations better model these phenomena, though at the cost of much higher processing runtimes. It was not possible to test the stages at choke since the booster stage was not able to overcome the test loop (piping) losses. From the trend of the tested curves (for both stages), it could be inferred that the stages choked earlier than predicted by the CFD analyses.

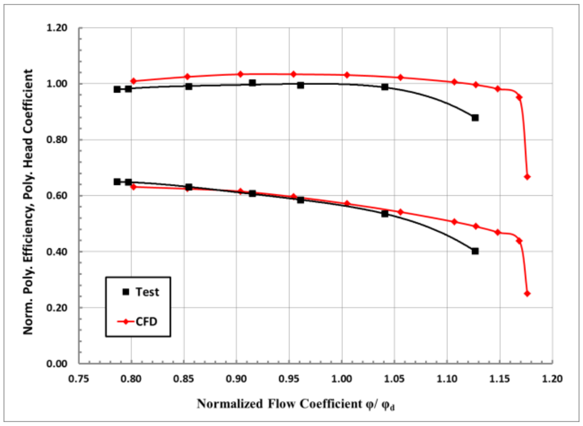

Figure 11: Stage 1 Performance – Test vs. CFD.

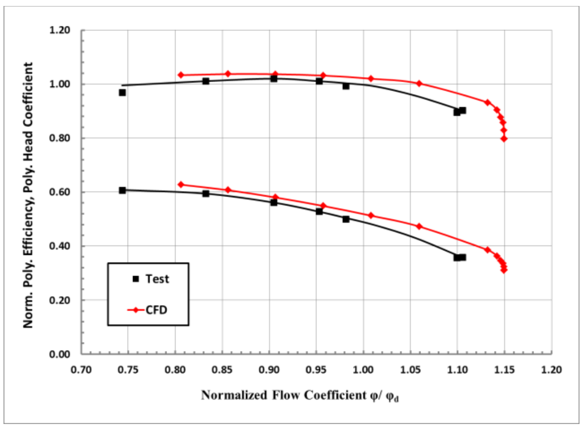

Figure 12: Stage 2 Performance – Test vs. CFD.

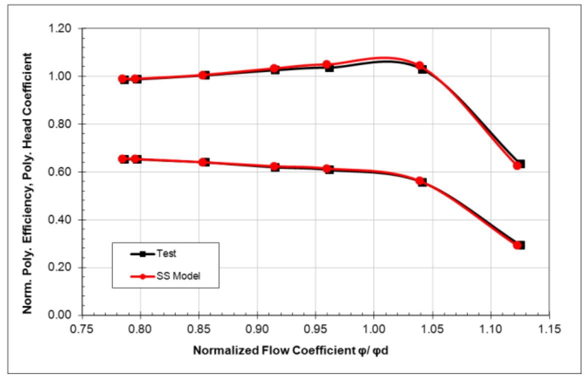

Figure 13: Section 1 Design Performance – Test vs. SS Model.

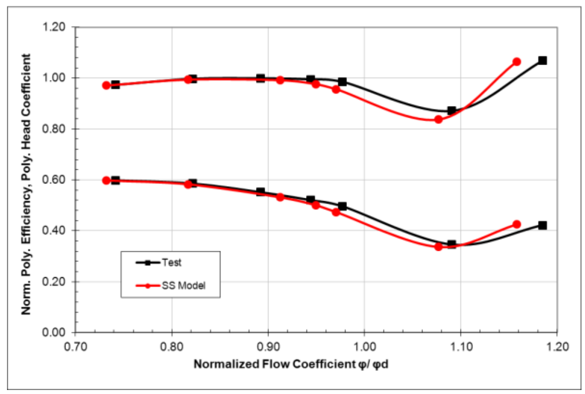

Figure 14: Section 2 Design Performance – Test vs. SS Model.

End users generally require that the centrifugal compressor meets API-617 [5] standard guarantees. These require that the compressor, without the use of a bypass, be suitable for continuous operation at any capacity at least 10% greater than the predicted surge capacity shown in the proposal, while there is no requirement for overload (However, many LNG operators generally require a 4% tolerance on overload margin). The overall power is required to be within +/-4%. The predicted stage performance was well within these tolerances.

Sidestream Modeling of Test Results

When testing sections with sidestreams, it is critical to maintain the flow function within a strict flow ratio tolerance. Failure to do so can result in significant variation of the sectional performance, as demonstrated and discussed by Fakhri et. al [3]. Although the PTC-10 code stipulates tolerances of ±5% on main inlet and ±10% on intermediate sidestreams, it was demonstrated that the use of such tolerances would result in substantial sectional performance variation [3], especially near overload flows. For this reason, the OEM's internal guidelines maintain the flow ratio within ±3%. Even at this tolerance some fluctuations exist. Near overload, it was not possible to be within this tolerance because of test loop limitations.

The sidestream model used the tested internal (stage) performance curves along with the test flow ratio and test inlet conditions to generate sectional performance curves. The first section was defined from main inlet flange to the sidestream flange. The second section was defined from the sidestream flange to the return channel exit of the second stage. In the case of sidestream sections, the sectional performance curves are "perceived" curves since the defined section does not constitute the full thermodynamic volume (separate flow enters the sidestream flange that is partially influenced by the preceding stage flow). These curves were then overlaid against the tested sectional curves. The normalized polytropic efficiency and head coefficient versus the normalized flow coefficient curves of these results are shown by Figures 13 and 14 for sections 1 and 2 respectively.

From Figure 13, it can be observed that there is good agreement between the sidestream model and test results. Throughout the operating range, sectional polytropic efficiency was within ±1.5% and the sectional polytropic head was within ±1.0%. The model and test demonstrated the non-smoothness in the sectional performance due to the flow function variation. The flow ratio for this sidestream requires a higher sidestream flow relative to the internal flow. As discussed by Fakhri et. al [3], this implies that the pressure at the sidestream flange will always be higher than the pressure at the exit of the return channel of the preceding stage. As a consequence, the sectional pressure ratio will always be higher than the internal (stage) pressure ratio – leading to a perceived sectional efficiency and head coefficient that is higher than the stage efficiency. If the flow ratio is maintained, the sectional efficiency will continue to rise and the sectional head coefficient will not drop as rapidly as the stage head coefficient (due to the rising sidestream flange pressure). This trend is shown in Figure 13. At the overload point, the flow ratio significantly decreased (i.e., lower relative sidestream flow) due to test loop limitations, leading to an abrupt drop in sectional efficiency and head coefficient.

Figure 14 shows the sectional curves for the second section. Similar to the first section, good agreement between the sidestream model and test curves can be observed. The sectional polytropic efficiency was within ±3.0% and the sectional polytropic head was within ±2.6%, throughout the operating range. Again, the non-smoothness of the sectional curves (caused by varying flow function) is observed. Since there is a pressure drop from the sidestream flange to the inlet pressure of the succeeding stage, the sectional pressure ratio will always be less than the stage pressure ratio at the design flow ratio. As a consequence, the perceived sectional efficiency and head coefficient will always be lower than the performance of the corresponding stage. This is demonstrated in Figure 14, with the exception of the overload point. As with the first section, test loop limitations forced a significant decrease in flow ratio, leading to lower total pressure at the flange relative to the inlet of the second stage due to the low sidestream flow. This resulted in the spike in sectional efficiency and head coefficient for that point. Along with a slightly higher error (relative to section 1 results), a flow shift is also noticed in the sidestream model curve relative to the test curve. This results from the sidestream model predicting a slightly lower mass flow to satisfy the test flow ratio. These errors, although minor, are likely compounded in each succeeding section due to factors not accounted for by the sidestream model (e.g. 1-D modeling vs. actual 3-D affects the flow is subjected to, impurities within the test loop affecting the mole weight of test gas, inaccuracy of real gas models, etc.).

As previously discussed, the machine was also run at off-design conditions, primarily to gather test data on internal performance. Both sections were tested simultaneously. For this reason, and also due to test loop limitations, the flow function was not maintained within strict tolerance at high flow levels. As before, the internal (stage) curves were used by the sidestream model along with the test flow ratio and external (sectional) results were generated. These were validated against the test results. Figures 15 and 16 show the low speed (95% design speed) performance results of the sidestream model vs. test for sections 1 and 2 respectively.

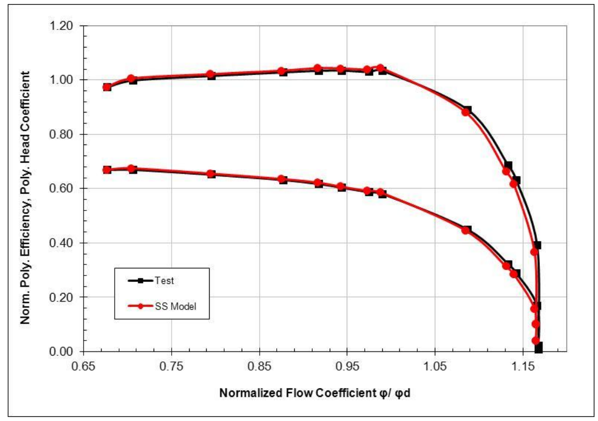

Figure 15: Section 1 Performance 95% Speed – Test vs. SS Model.

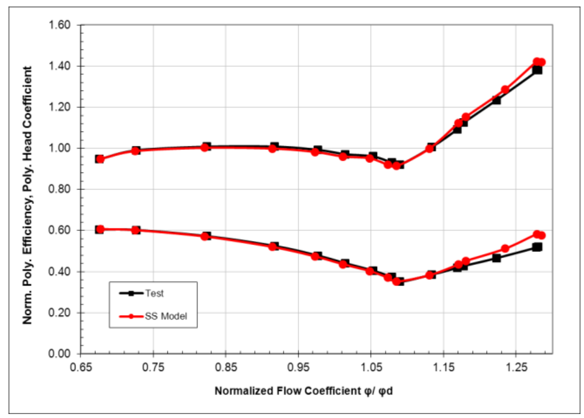

Figure 16: Section 2 Performance 95% Speed – Test vs. SS Model.

There is good agreement between the sidestream model and test results for the first section, as seen in Figure 15. There is more unevenness in these curves than that observed in Figure 13, due to the higher variation of the flow function. Throughout the operating range, the sectional polytropic efficiency was within ±2.0% and the sectional polytropic head coefficient was within ±2.0%. Due to the decreasing flow ratio (i.e. lower sidestream flow) with higher flow, perceived sectional efficiency and head coefficient are also observed to fall precipitously.

Figure 16 shows that there is also good agreement between the sidestream model and the test results for the second section. Since the flow ratio significantly decreased when the machine was operating close to stage two overload, the sectional efficiency is observed to be quite high at these flows. Lower flow ratio implies lower sidestream flow, leading to a flange total pressure that is lower than the total pressure of the mixed flow at inlet of the second stage. Hence, the lower flange total pressure leads to the perceived sectional pressure ratio being actually higher than the stage pressure. This, in turn, causes the perceived sectional performance to be higher than the stage performance, which is what is observed in Figure 16 at the higher flows. Once the target flow ratio is attained, the perceived sectional performance shows similar trends as those observed in Figure 14. Throughout the operating range, the sectional polytropic efficiency was within ±3.0% and the sectional polytropic head was within ±3.0%.

Movable guide vanes just upstream of the inlet to each stage allowed pre-whirl to be added to the flow entering the respective stages. The primary purpose of adding pre-whirl to inlet flow is to flow shift the internal performance curve, allowing the stage to produce a higher/lower head at the same inlet volume flow than what would have been possible with radial incoming flow. High "against" pre-whirl of 7.5 degrees and 10 degrees was added to the first and second stages respectively. The rationale for this was that once performance is tested with no pre-whirl and at this high pre-whirl setting, performance for any other intermediate pre-whirl setting can be estimated within reasonable accuracy. Similar to the low speed test, both sections were tested simultaneously and the flow ratio near overload was not strictly maintained. Figures 17 and 18 show the performance of both sections with pre-whirl added to their respective stages.

Figure 17 shows good agreement between the sidestream model and test results for the first section. Similar to the performance curves at 95% design speed, non-smoothness in the curves is observed at high flows, due to the higher variation of the flow function. Throughout the operating range, the sectional polytropic efficiency was within ±1.5% and the sectional polytropic head coefficient was within ±1.5%. Similar to the low design speed test, the decreasing flow ratio at higher flows causes a decrease of sectional efficiency and head coefficient.

Figure 18 shows good agreement between the sidestream model and test results for the second section. Once again, the decreasing flow ratio (i.e. lower sidestream flow) at higher flow results in a sidestream flange total pressure that is lower than the second stage inlet pressure, leading to a high perceived sectional efficiency and head coefficient. At design flow ratio, the trend in the performance reverts back to that observed in Figures 14 and 16. Throughout the operating range, the sectional polytropic efficiency was within ±2.0% and the sectional polytropic head was within ±3.0%.

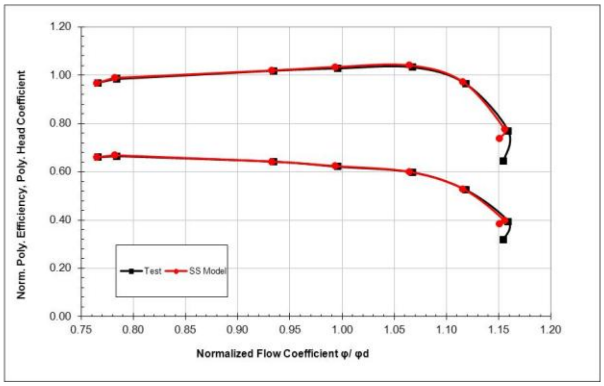

Figure 17: Section 1 Performance Pre-whirl (-7.5 deg.) – Test vs. SS Model.

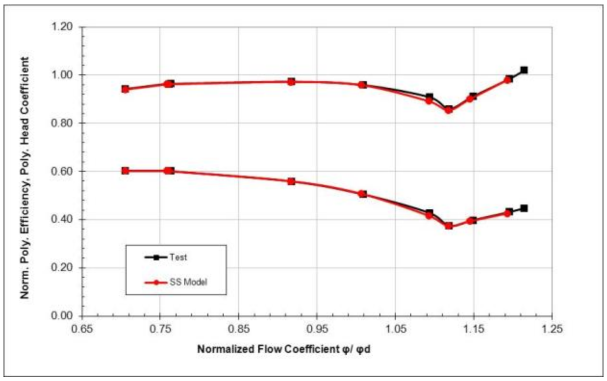

Figure 18: Section 2 Performance Pre-whirl (-10 deg.) – Test vs. SS Model.

Since the sidestream model was validated by these test results, it could be used to accurately model the performance of a multiple section, multiple sidestream machine with good confidence. Therefore, the sidestream model was used to predict the performance of the actual production machine. As discussed in previous sections, the propane production machine consisted of four sections and five stages with three sidestreams. All sidestreams were designed per the criteria discussed. The last three stages were contour trims of the second stage. Therefore, the tested internal (stage 1 and stage 2) curves of the large scale validation compressor (design speed, low speed and against pre-whirl) were used by the sidestream model to predict the performance of the five-stage production machine with three sidestreams. The sidestream model was run at constant design flow function and stable inlet conditions (pressure, temperature, speed) for all three sidestreams. The results shown in Table 1 use the stage performance with pre-whirl and shaft speed reduced to achieve discharge pressure. Table 1 presents the predicted performance compared with the client specified tolerances. Only the maximum and minimum performance deviations from all four sections are shown in the table. Most of the critical tolerances were easily met with the exception of certified speed (-1.3%), certified head (-1.7%) and choke capacity (-2.7%). These tolerances could be easily met by slightly increasing the speed and reducing the pre-whirl with a small increase in shaft power (within tolerance).

Table 1: Predicted performance variation compared to client specified tolerances

Criteria	**Specified Tolerance**	**Performance Variation**
Shaft Power	< +2.0%	+0.9%
Certified Speed	-1.0%, +0.0%	-1.3%
Certified Head	-0.0%, +3.0%	-1.7%, +1.4%
Sidestream Pressure	-2.0%, +2.0%	-1.0%, +0.4%
Choke Capacity	> -2.0%	-2.7%, +3.2%
Stability (Surge/Stall)	> -2.0%	+2.4%, +6.8%

CONCLUSIONS

This paper discussed the successful testing and validation of a large scale, two-section LNG sidestream compressor. Each section consisted of a single high flow, high Mach stage with a sidestream in between the sections.

The performance of these stages was verified through CFD analyses. The OEM has in the past obtained good agreement between CFD and test for high-Mach stages [1]. On conducting the test at design conditions for the machine large scale validation compressor, the tested internal (stage) results agreed well with the CFD analyses, lending further credibility to the accuracy of CFD analyses.

Three separate tests were conducted on the machine: (1) design speed for sections 1 and 2; (2) 95% speed for both sections; and (3) against pre-whirl flow added to both stages. The results from all three tests were discussed and the sidestream model was validated against them. It was shown that the sidestream model could accurately predict sectional performance, even when test inlet conditions and flow ratio vary. The sidestream model was then used to predict performance of the five-stage, three-sidestream propane production compressor. The comparison of the test results (with pre-whirl) and the tolerance requirements was shown in Table 1. Most of the criteria were met; and with a slight pre-whirl adjustment and speed increase all the criteria would be met. Inlet guide vane and speed adjustments are commonly used approaches to tune compressor performance.

Similar to the conclusions drawn previously by Fakhri et. al [3], it was observed that a variation in flow ratio can significantly affect sectional performance of compressors with sidestreams. Current ASME PTC-10 guidelines stipulate that OEMs maintain flow ratio for sidestream sections within ±10% when testing. This tolerance on flow ratio will not often meet client requirements or even API-617 requirements on sectional performance – especially for compressors where sections contain few stages of low pressure ratio and high sidestream flows. In such cases, sidestream flange pressures will vary extensively. This implies that the external (sectional) pressure ratio can be significantly different from the internal (stage) pressure ratio – resulting in a large fluctuation of sectional polytropic efficiencies and head coefficients. For these reasons, the OEM ensures that the flow ratio is maintained within ±3% when testing sidestream sections.

It is recommended that the requirement for allowable flow ratio fluctuation in the ASME PTC-10 code be made more stringent for sections with incoming sidestreams to a minimum of ±5%. This would generally ensure that the sectional performance variation would stay within acceptable bounds. Despite this, even this lower flow ratio variation will still result in some unevenness of the sectional curves. This could be corrected by using the sidestream model and these corrected curves could then be used to determine if client requirements have been met. Since sidestream machines used for processes (such as LNG refrigeration processes) require fixed sidestream flange pressures, it is suggested that the flow ratio be varied based on inlet flow to achieve the required sidestream flange pressures.

Nomenclature

ϕ = flow coefficient = $700.33 \dfrac{Q}{N D_2^{\,3}}$

A0 = sonic velocity of gas in feet per second

D_2 = impeller exit diameter in inches

N = operating speed in rotations per minute (rpm)

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